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Plate heat exchangers: Recent advances

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ABSTRACT

This study presents the advances in plate heat exchangers both in theory and application. It dresses the direction of various technical research and developments in the field of energy handling and conservation. The selected areas of heat transfer performance and pressure drop characteristics, general models and calculations change of phase; boiling and condensation, fouling and corrosion, and welded type plate heat exchangers and finally other related areas are highlighted.

Contents

1. Introduction

Plate heat exchanger (PHE) is now commonly used in a wide range of chemical process and other industrial applications with a particular attention from the food industry due to several reasons such as: suitability in hygienic applications, ease of cleaning and the thermal control required for sterilization and pasteurization. Also PHEs exhibit excellent heat transfer characteristics which allow more compact designs than achievable with conventional shell and tube heat exchangers, and have a very large surface area in a small volume and can modified for different requirements simply by increasing or decreasing the number of plates needed. With these advantages, along with advances in material technology in the form of new temperature- and pressure-resistant materials for gasket or graphite plates, it is now possible to use this class of heat exchangers appropriately for the power and chemical processes.

Even though plate heat exchangers are mostly used in liquid-liquid heat transfer duties which require uniform and rapid heating or cooling. But there is an increase trend to use PHEs in the evaporation and condensation duties for plant energy conversion.

On the other hand, the main disadvantage of PHE is the limit of its operational range where the maximum operating pressure is limited to 20.4 bar and the operating temperature to about 150 °C. These operational conditions can be extended to about 40.8 bar and

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800 °C in lamella type PHE which does not have the flexibility of the gasket plate unit.

Plate heat exchangers can be fabricated in gasketed, welded, or module welded design characterized by the model in which the flow channels for the two heat exchanging media are sealed. According to the type of heat exchanger the individual plates are sealed relative to each other by gaskets placed in circumferential grooves or by welding.

Plate heat exchangers are first fully described in [1], and there are several comprehensive compiled materials on various design aspects in the literature [2–7].

The main objective of this review is to highlight the recent advances which affects the performance of plate heat exchangers specially in the industrial section. As an industrial application, Karlsson [8] evaluated the performance of plate heat exchangers in residential water radiator heating systems receiving their heat from geothermal resources. Recent experimental and numerical work to analyze the flow in an oil/water plate heat exchanger for the automotive industry was conducted [9]. Also, the use of plate heat exchangers to improve energy efficiency in phosphoric acid production was illustrated [10]. Downsized exchanger without loss of thermal-hydraulic performance is crucial matter for the industry applications [11]. The improvement of compactness is a vital issue carried by more competitive surface shape under the carefully designed riblet angle [12]. The use of nanofluids as coolants in industrial heat exchangers seems inauspicious [13], and the only drawbacks so far are the high price and the possible instability of the nanoparticle suspensions [14].

2. Thermal & hydrodynamic characteristics

From the early literature on the effect of plate arrangements on flow distribution and pressure drop was presented by Bassiouny [15,16]. On the other hand, Thonon and Mercier [17,18] presented an overall design method used for sizing plate heat exchangers. The method is based on the temperature enthalpy diagram, and introduced a model taking into account flow maldistribution effects for single and two phase flows [19]. The effects of flow maldistribution was presented through a general thermal model in terms of Effectiveness-Ntu and LMTD relationships.

Rao et al. [20], showed in details the effect of flow maldistribution and presented a wide range of parametric study which brings out effects such as those of the heat-capacity rate ratio, flow configuration, number of channels and correlation of heat transfer. Also, the experiments showed the effect of pressure drop on flow maldistribution [21]. Noninvasive technique of Positron Emission Particle Tracking (PEPT) was used to investigate the flow pattern in a plate heat exchanger [22]. Recently, Tsai et al. [23] investigated hydrodynamic characteristics and distribution of flow in two cross-corrugated channels of plate heat exchangers. Effects of dissipation and temperature-dependent viscosity on the effectiveness calculation was addressed by Gherasim et al. [24].

Martin [25] developed the generalized Lévêque equation – a theoretical equation – to predict the plate heat exchanger thermal performance. Also, Dovic et al. [26] developed generalized correlations for predicting heat transfer and pressure drop which are used to predict the performance of chevron-type plate heat exchangers by obtaining the heat-transfer coefficients in fully developed laminar or turbulent channel flow. Dumas and Corradini [27] analyzed the influence of the thermal resistance of the fluid film, temperature distribution profiles and enthalpy efficiency in the range of temperatures normally used in civil applications. Ciofalo [28] Explored the effect of the longitudinal heat conduction along the dividing walls and showed that it may enhance the exchanger's performance.

2.1. Influence of plate types & configurations

Extensive experimental works were conducted on different chevron type plates to study the effects resulted from the variation of parameters such as: pitch, amplitude, and chevron angle. This is to understand their influence on heat transfer and flow patterns [29,30]. Heat transfer and isothermal pressure drop data for single-phase water flows in a single-pass U-type counter-flow PHE and in low Reynolds number flows are presented through the use of different chevron plate arrangements: two symmetric plate arrangements with beta = 30°/30° and 60°/60°, and one mixed-plate arrangement with beta = 30°/60°. Also, the effects of chevron angle beta in these three different plate arrangements were illustrated. The impact of corrugation aspect, ratio gamma, and flow conditions on Nusselt Number (Nu) and friction factor (f) characteristics were outlined [31–38].

The size (i.e., height and pitch) of the corrugation embossed on the plates, and the orientation of the corrugation with respect to the main flow direction on the heat transfer performance of the exchanger were investigated [39]. Charre et al. [40] presented a general heat transfer and pressure drop model which is based on the theory of porous media and included the influence of 11 geometric parameters of the plate. A new-type corrugation Plate was designed where t the flow resistance of the working fluid in this new corrugation PHE, compared with the traditional chevron-type one, was decreased by more than 50% [41]. The laminar flows of Newtonian and power-law fluids through cross-corrugated chevron-type plate heat exchangers (PHEs) were numerically studied in terms of the geometry of the channels [42].

The influence of grooves in the U-turn areas for the multichannel-plate heat exchangers (MCPHEs) was investigated by Chang et al. [43] using of acrylic plates. Plates with dimples [44] were designed to enhance heat transfer and reduce fouling. Various shapes of rib-roughened surfaces, different rib spacing and rib arrangements were applied to the wider walls of the duct to enhance the heat transfer in a plate heat exchanger [45].

Pinto and Gut [46] and Gut [47,48] developed an optimization method for determining the best configuration(s) of gasketed plate heat exchangers, and their objective was to select the configuration(s) with the minimum heat transfer area that still satisfies constraints on the number of channels, the pressure drop of both fluids, the channel flow velocities and the exchanger thermal effectiveness. A general method for the optimal design with undulated surfaces was proposed by Kanaris et al. [49] and Arsenyeva et al. [50]. Recently, exergy analysis was included as an important variable in the design procedure [51,52].

2.2. General procedure calculations

General calculation procedure for plate heat exchangers and useful charts were developed [53], in terms of the number of transfer units (Ntu) and the heat capacity rate ratio (R), for 150 plate heat exchanger configurations. These exchangers were classified on the basis of number of channels, number of passes of each fluids and flow arrangement. Specific guidelines for selecting the appropriate plate heat exchanger configuration were proposed. Wright and Heggs [54,55] calculated the effectiveness of a single pass two stream plate heat exchanger (PHE) when one stream undergoes a phase change; specifically condensation, and presented analytical solution for the system under the assumption of constant overall heat transfer coefficient when run in either co-current or counter-current arrangements. Also the authors extended their analysis to systems in which the overall heat transfer correlation is dependent upon the quality of the phase change stream. Recently, Lin et al. [56] derived dimensionless correlations using the Buckingham Pi theorem to characterize the heat

transfer performance of the corrugated channel in a plate heat exchanger.

2.3. Heat transfer coefficient measurements

Both transient and electrochemical mass transfer techniques were widely used for the measurement of heat transfer coefficient in heal exchangers. Also constructing the flow regime map was another useful method for the measurement of HTC. Roetzel [57] experimentally evaluated thermal parameters of plate type heat exchangers using a temperature oscillation technique, and a mathematical model with axial dispersion was utilised to evaluate heat transfer coefficient and dispersion coefficients characterized by Number of transfer units (Ntu) and Peclet number respectively.

Ros et al. [58] applied the transient-state technique to measure the global heat exchange coefficient between a liquid and corrugated plates, and modelled the fluid flow by an equivalent flow pattern obtained by inert tracer experiments. The authors used the frequency response to estimate the heat transfer coefficient between the fluid and the solid. Quarini et al. [59] illustrated the local heat transfer characteristic of an APV junior paraflow plate heat exchanger. Heggs et al. [60] and Heggs and Walton [61] employed an electrochemical mass transfer technique to calculate values of the local transfer coefficients within a corrugated plate heat exchanger channel for limited range of Reynolds number from 150 to 11,500 for the following corrugation angles: 30, 45, 60 and 90. The authors presented mass transfer profiles on both sides of the channel and proved that the peak in mass transfer at the base of the corrugation was consistent with a swirling motion which dependent upon the channel flow rate.

Ciofalo [62] obtained the distributions of the local heat transfer coefficient by using liquid-crystal thermography, and computed the surface-averaged values and measured friction coefficients by wall pressure tappings. Also drived overall heat transfer and pressure drop correlations. Whereas, Vlasogiannis et al. [63] measured the heat transfer coefficient of air/water mixture – the cold stream – as a function of air and water superficial velocities by constructing of a flow regime map using high-speed video camera for a plate heat exchanger under two-phase flow conditions. A new plate heat exchanger for water-refrigerant systems such as chillers was developed. Plates embossed with pyramid-like structures were stacked up to form the heat exchanger [64]. The measured heat transfer coefficients of the plates (convective vaporization) were about one and a half to two times higher than those of commercial herringbone-type plate heat exchangers. Recently, Freund and Kabelac [65] developed a method to measure local convective heat transfer coefficients using temperature oscillation IR thermography and Computational Fluid Dynamics (CFD).

2.4. Numerical and analytical models

Numerical models were solved using different methods such as: 3-D finite volume technique which was used to study the effects of flow channel angles and cross-sectional shapes of exchanger plates. This is to determine the optimum design parameters for the exchanger [66,67]. Whereas, Rebholz et al. [68] implemented a 2D finite volume technique for the prediction of laminar flow, and presented solutions for the end effect of plate heat exchangers for changing flow conditions in multipass arrangements and several different configurations. Heggs and Narataruksa [69,70] used two numerical schemes: a shooting method a fourth-order Runge-Kutta method and central finite-difference method to obtain solutions of PHE thermal performance. But the shooting method was only applicable for single pass looped flow arrangements. Fiebig et al. [71] numerically analyzed heat transfer and flow loss with longitudinal vortex generators as fins.

There were an extensive investigations for counterflow plate heat exchangers based on a dispersion model which took the deviation from ideal plug flow into consideration to predict the response due to temperature transients. The 'phase lag effect', which is a special characteristic of plate exchangers, played a significant role in the dynamic regime [72,73]. The axial dispersion reduced the exergetic performance. The minimum irreversibility corresponding to a given level of dispersion was identified. A new concept by analogical treatment of hyperbolic axial dispersion with the fluid conduction was introduced, and it takes the flow maldistribution into account in the analysis of heat exchangers [74]. Furthermore, Das and Roetzel [75] improved the conventional axial heat dispersion model by considering dispersion as a wave phenomenon propagating with a finite velocity. Strelow [76] proposed a general calculation method to simulate the heat flux along the walls of the plates as well as the dispersion in the passages.

Bigoin et al. [77] and Miura et al. [67] used the Computational Fluid Dynamics (CFD) method which is based on the numerical simulation of the turbulent flow using various turbulence models (mixing length model, eddy viscosity model and large eddy simulation) and various meshes. The simulations of stirred yoghurt processing in a plate heat exchanger were performed using computational fluid dynamics (CFD) calculations. CFD program is used to evaluate the tortuosity coefficient which is used to estimate Fanning friction factors and convective heat transfer coefficients [78–80]. Whereas, Kanaris et al. [81] explored the potential of using a general purpose CFD code to compute the characteristics of the fluid flow and heat transfer augmentation in conduits with corrugated walls encountered in commercial plate heat exchangers. Parallel and series flow arrangements were tested and experimental results were compared to numerical predictions for heat load obtained from the 3D CFD model and also from a 1D plug-flow model [82]. Recently, it was found that the use of depth-averaged flow and energy equations reduced the elapsed time of CFD simulations [83]. Also, a simplified numerical simulation to obtain correlations for the determination of convective heat transfer coefficients of stirred yoghurt during the cooling stage in a plate heat exchanger [84]. Simulation of the three-dimensional temperature, pressure, and velocity fields were obtained [85]. Effectiveness charts were generated for counterflow arrangements using computational fluid dynamics (CFD) method [86].

On the other hand, Mehrabian [87] and Mehrabian and Poulter [88] developed analytical solutions for temperature distributions within a plate heat exchanger, and studied uniform heat flux, constant overall heat transfer coefficient (U), linearity between (U) and Temperature (T), and linearity between (U) and Delta (T). The work was extended to focus on experimental approach for local pressure and local temperature measurements to understand the hydrodynamic and thermal characteristics of corrugated channels.

Ho et al. [89–91] studied analytically the influence of recycle on a parallel-plate heat exchanger of inserting in parallel an insulation sheet to divide an open duct into two channels for double-pass operations with uniform wall temperature. Effects of variable ratio of heat fluxes on both sides and impermeable-sheet location were also studied [92]. For laminar flow with counterflow parallel-plate heat exchangers, Vera and Linan [93] provided a solution for the temperature field. The solution involved eigenfunction expansions that were solved in terms of Whittaker functions using standard symbolic algebra packages leading to analytical expressions that provided the eigenvalues numerically.

The dynamic behaviour was studied and evaluated either through a temperature step input to generate the temperature profiles along the channels and in the outlets [94] or a step flow variation conducted by Dwivedi and Das [95] through predictive model which included the effect of the port to channel maldistribution on the performance of plate heat exchangers. Das and

Murugesan [96], Srihari and Das [97], Shaji and Das [98] presented an analysis to predict the transient response of multipass plate heat exchanger based on an axial heat dispersion model in the fluid which takes deviation from ideal plug flow into consideration.

3. Two phase systems

Boiling heat transfer was investigated in both sub-cooled and saturated flow boiling modes. Polat et al. [99] looked into the forced convective boiling of a non-newtonian liquid in a multipass plate heat-exchanger. Hsieh et al. [100] investigated experimentally the sub-cooled flow boiling heat transfer characteristics of refrigerant R-134a and demonstrated in details the effects of the boiling heat flux, refrigerant mass flux, system pressure. Furthermore, Hsieh and Lin [101,102] conducted experiments on saturated flow boiling heat transfer and the associated frictional pressure drop of the ozone friendly refrigerant R-410A and established empirical correlations for the saturated boiling heat transfer coefficients and friction factor in terms of the boiling number and equivalent Reynolds number.

Whereas, Andre et al. [103], evaluated heat transfer in the evaporation of ammonia in a plate heat exchanger. Experimental results on evaporation heat transfer for flow boiling of ammonia and of R134a in a chevron-pattern corrugated plate heat exchanger (PHE) were presented by Djordjevic and Kabelac [104]. From these results, it was shown that the parallel flow case yields better overall performance than the counterflow case, and that plates with low chevron angle corrugations increased the evaporation heat transfer. Recently, Cerezo et al. [105] and Taboas et al. [106] conducted experimental work on the saturated flow boiling heat transfer and associated frictional pressure drop of ammonia/water mixture flowing in a vertical plate heat exchanger. Also, boiling characteristics for other solutions were evaluated such as $LiBr–H_2O$ and $NH_3–H_2O$ [107], $NH_3–LiNO_3$ and $NH_3–NaSCN$ solutions [108], and tetrabutylammonium bromide (TBAB) clathrate hydrate slurry (CHS) as a secondary refrigerant [109]. The effect of boiling two-phase flows and different flow pattern were visualized by thermal neutron radiography method to study their effect on the heat transfer performance [110].

Plate heat exchangers started to play an important role in the industrial operations used as evaporators or condensers. Two phase flow research was extensively conducted [111–113]. The most common two phase system is air–water. The flow characteristics were addressed including flow pattern and pressure drop inside a plate heat exchanger. The overall pressure drops of low and medium chevron angle configurations were found to be independent of channel gap, while the heat transfer section results showed a considerable influence for isothermal air/water two-phase flows [114,115]. The correlations to predict heat transfer coefficients for boiling and condensation in a particular brazed plate heat exchanger were established by Hickson [116].

For brazed plate heat exchangers, Bogaert and Bolcs [117] and Jokar [118], defined the thermal and hydrodynamic performances in terms of the hydraulic diameter parameter, and developed only one equation predicting hydrodynamic and thermal characteristics in the turbulent and laminar-transitional flow regions. Whereas, Wang et al. [119] obtained the heat transfer and pressure drop characteristics of complete steam condensation and partial condensation. Ayub [120] presented a literature survey and new heat transfer and pressure drop correlations for refrigerant evaporators. Due to the high efficiency and compactness of brazed plate heat exchangers, the condensation and vaporization of high pressure refrigerant fluids were implemented as evaporators and condensers in chiller and heat pumps. Both HC and HFC refrigerants were recently studied by several researchers due to their environmental impact as shown in Table 1. Garcia-Cascales et al. [121] studied the refrigeration cycles in which plate heat exchangers were used as either evaporators or condensers. Also, several heat transfer coefficients were evaluated in the refrigerant side for R-22 and R-290. Recently, Hayes et al. [122] conducted experimental investigation of carbon dioxide condensation in brazed plate heat exchangers.

Table 1
Various refrigerants investigated.

Researcher	Refrigerant investigated
Dutto et al. [123]	HCFC-142b
Pelletier and Palm [124]	HCFC-22
Kedzierski [125]	HCFC-22
Yan et al. [126,127], Yan and Lin [128]	HFC-134a
Palmer et al. [129]	HCFC-22
HC-290, HC-290/HC-600a (70/30 wt%), HFC-32/HFC-152a (50/50 wt%))	
Thonon and Bontemps [130]	HC-601, HC-600,
HC-290	
HC-600/HC-290 ((28/72 wt%)	
and (49/51 wt%))	
Longo et al. [131]	HCFC-22
Kuo [132],	
Kuo et al. [133]	
Longo and Gasparella [134], Longo [135]	HFC-410A
Park and Kim [136]	
Jassim [137],	
Longo [138],	
Longo and Gasparella [139,140]	HFC-134a
Longo [141]	HC-600a,
HC-290	
HC-1270	
Longo [142]	HFC-236fa
HFC-134a	
HFC-410A	
Longo [143]	HFC-600a
HFC-290	
HFC-1270	

4. Fouling & corrosion

The significance of fouling phenomena came from the fact that fouling deposits increase the thermal resistance to heat flow. The fouling has an extremely complex behaviour and this was one of the main reasons why plate and frame heat exchangers are not widely installed in the chemical process industry. Fouling results in hydraulic and thermal disturbances and creates the need for cleaning operations which have to be carried out to bring the exchanger surface back to its original state. Fouling in the food industry is a severe problem compared with other industries. It reduces PHE efficiency, food quality and can also give rise to microbiological problems.

There are several types of fouling mechanisms were documented in the literature, and for liquid-side fouling they are:(1) precipitation and crystallization fouling [144–148], (2) chemical reaction fouling, (3) particulate fouling [149], (4) corrosion fouling (5) biological fouling, and finally (6) solidification and freezing fouling. Changani et al. [150] presented a review describing research into both the engineering and the chemical factors that lead to deposition of protein and minerals on the plate surfaces. Also, Visser and Jeurnink [151] reviewed the main factors in the fouling of processing equipment used for heating dairy fluids

Fouling in plate heat exchangers is function of the plate geometry and fluid velocity. Thonon and Grillot [152], Bossan [153] and Bossan et al. [154] illustrated the evidence of an asymptotic fouling resistance behaviour, and presented an attempt to account for the influence of the flow maldistribution between the heat exchanger channels. Also, various corrugation patterns were fouled

at different rates under identical process conditions, and that these differences were attributed to the effects of flow distribution on fouling rates in the plate channels [155]. Karabelas et al. [156] reported new fouling data for plate heat exchanger of two angles of corrugation, (30 and 60°) and particles of mean size ∼5 μm and their economic implications. The effect of surface energy, wettability and surface roughness on the deposition of calcium sulphate on plates were studied [157].

In the dairy industry, milk processing is considered a major problem causing fouling of plate heat exchangers. Mathematical modelling and simulation of complex plate heat exchanger arrangements under milk fouling were carried out using detailed dynamic models. Georgiadis et al. [158] and Georgiadis and Macchietto [159] and Puhakka et al. [160] proposed complex fouling models based on either reaction and/or mass transfer scheme coupled with a general thermal dynamic model of plate heat exchangers. Advanced work was presented by constructing a 2D dynamic fouling models for milk fouling. These models showed that the aggregation rate of unfolded protein was found to increase exponentially with increasing wall temperature and was accompanied by a substantial reduction in the heat-transfer coefficient [161,162]. Carezzato et al. [163] illustrated experimental data obtained from non-Newtonian heat transfer for eight different configurations.

Robbins et al. [164] compared the fouling from whey protein concentrate (WPC) and milk in a plate heat exchanger. Whereas, Christian et al. [165] investigated the effect of adding minerals (calcium and phosphorus) on fouling and cleaning behaviour of Whey protein concentrate. The effects of fouling by whey proteins on several flow arrangements of a plate heat exchanger equipped with straight corrugation plates using the measurement of both the overall heat transfer coefficient and the dry masses of deposit were studied [166,167]. The influences of calcium concentrations, Reynolds number and temperature played an important role on the deposit structure and the rate of growth of whey protein deposition in a plate heat exchanger [168]. An antifouling coating with low surface energy (low wettability) led to a hydrophobic and oleophobic effect. Coating stainless steel plate surfaces with commercially available food-grade materials; Lectrofluor-641™, graded Ni-P-PTFE, and AMC148-18 was one option to be used for possible thermal energy savings in food processing equipment [169]. Polyurethane coated plates using nano-composites coatings had shown to reduce considerably fouling inside gasketed plate heat exchangers used in milk production [170].

Monitoring of fouling formation is another direction of research. The on-line monitoring to verify assumptions regarding heat exchanger fouling, strainer design, and material compatibility was considered by Nolan and Scott [171]. They used side stream monitor (SSM) as a useful test platform for the selection and optimization of a chemical treatment program for control of bio-fouling in raw water service. Also, the EAF technology was developed for the purpose of mitigating scales in both plate-and-frame and shell-and-tube heat exchangers by Cho and Choi [172]. Whereas, Rivero and Napolitano [173] described a practical procedure based on artificial neural networks (ANN) that allowed the prediction of the deposit thickness, the overall heat transfer coefficient and the critical time for reducing the impact of fouling on Pasteurization processes. Recently, Merheb et al. [174] proposed a new acoustic technique to monitor fouling inside PHE in real time.

Plates are normally manufactured in stainless steel as a standard material which was experimentally studied for the liquid-pase particulate fouling [175]. But titanium and aluminum brass are commonly used. As a result of the technology, the surface improvement of aluminum alloy specimen was achieved without thermal degradation and surface treatment to enhance the corrosion resistance [176].

In practice, the most common types of localized corrosion observed on plate heat exchangers were pitting [177], crevice corrosion, and stress corrosion cracking. Turissini et al. [178,179] and El-Batahgy [180] discussed the corrosion failures in plate heat exchangers, and studied both the effect of crevice corrosion under gasket and stress corrosion cracking to cause failure to a plate heat exchanger. Singh et al. [181] investigated the causes of the gasket failures and implemented process control solutions with limited success. Then examined the factors in determining a suitable alternative gasket material specification.

5. Welded plate heat exchangers

Welded plate heat exchangers have wider range of use than gasketed plate heat exchangers where the operating temperature range from −50° C up to 350° C and operating pressures from full vacuum to 40 bar. Chopard et al. [182] illustrated how the compact technology was developed with the technique of welded or soldered plates. The developed welded plate heat are capable to overcome the pressure and limitations of gasketed plate-and-frame exchangers. Blomgren [183] thoroughly described the structure of a welded plate heat exchanger where the edge portion of each heat transferring plate was welded together with the edge portions of a first adjacent heat transferring plate along an outer line and with the edge portion of a second adjacent heat transferring plate along an inner line. Reppich [184] developed a laser welded modular design of a plate heat exchanger to handle aggressive media. The design kept the inherent advantages of plate type heat exchanger where it can be disassembled and mechanically cleaned outside the modules. Zhu and Liao [185] conducted experiments for heat transfer and pressure drop of water flowing in the all-welded plate heat exchangers with various plate numbers, plate width and plate length. Recently, the heat transfer and pressure drop characteristics of welded type plate heat exchangers for absorption application using Computational Fluid Dynamics (CFD) technique was examined and showed that the plate with the elliptical shape gave better performance than the plate of the chevron shape [186].

6. Other related areas

In plate heat exchangers, compensating for end effect is critical issue in the performance calculations and was addressed by Polley and Abu-Khader [187]. Plate-fin heat exchangers are categorized as a compact heat exchanger due to its relatively high heat transfer surface area to volume ratio. They are mostly used for low temperature services such as natural gas, air separation plants and aerospace industry. The designs include crossflow [188–190] and counterflow coupled with various fin configurations. Recent review on plate-fin exchangers was presented by Sheik Ismail [191]. An extensive works on thermo-hydraulic models [192,193], design methods [194], plate geometry and fin type effect [195–197], sizing of multistream plate-fin exchanger [198–203], two phase flow [204,205], particulate fouling effect [206], cost optimization [207] and numerical and CFD simulations [208–212] were investigated.

7. Conclusions

The selected areas discussed in this review are ones had more attention in last decade. Further research in different aspects in these areas can be suggested and extra work can be carried out. Some of these ideas which need developing and polishing such as: (a) compactness and downsized exchanger without the loss of thermal-hydraulic performance which is a crucial matter for the industry applications, (b) theoretical development of the he Danilova equation and the Steiner boiling correlation adapted to

PHEs, (c) still there is a strong need for proposing further techniques to reducing fouling in food processing equipment which will have a direct impact on operational cost, and last and not least (d) the use of nanofluids and their role in the design aspects of the exchanger, which is considered a new growing research area. As plate heat exchangers are going more and more into sever process conditions, corrosion of plates is crucial problem facing the industry due to high operational cost paid.

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